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NUMERICAL-EXPERIMENTAL ANALYSIS OF PLATE-FINNED HEAT EXCHANGER WITH DIFFERENT FIN GEOMETRY: INFLUENCE OF THERMAL CONTACT AND TURBULENCE PROMOTERS IN NUSSELT NUMBER

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Abstract. *The present work discusses the effects of delta winglet vortex generators and thermal grease on a heat exchanger with perforated plate fins based on a hybrid numerical-experimental analysis. This study compares the heat transfer rate for the same system with and without the addition of the vortex generators and thermal grease on the thermal contact between tubes and fins. The system is composed of four heaters connected by three perforated plate fins, made of aluminum. The hybrid methodology used in this paper consists of experiments performed on a wind tunnel, from which the boundary conditions for the numerical simulations were obtained. The commercial program Ansys-Fluent® was adopted so solve the governing and turbulence equations. The methodology is validated through comparisons with results from literature. The main results indicate an increase of the Nusselt number close to 26% when vortex generators are used, and an average of 12% when thermal grease is used to improve the contact between tubes and fin.*

Keywords: *Finned heat exchangers, wind tunnel, hybrid methodology, vortex generators, Nusselt number*

1. INTRODUCTION

Finned tube heat exchanges are widely employed in industry, household and commercial applications, due to their high efficiency, simple construction and relatively lower cost (Rauber et al., 2022). As a result of this demand, it is of great interest to use different techniques to improve performance of these equipment, always aiming to maximize the heat rate and minimize the pressure drop (Zanatta et al., 2020). In this context, the recent literature presents several works describing enhancement techniques as well as application to different cases.

The fins are added to the heat exchanger in order to augmented the heat transfer area, such as the works of Chen, Chou and Wang (2007), Huang, Yuan and Ay (2009), Choi et al. (2010), Wang et al. (2019) and Rauber et al. (2022), who studied the effect of some heat exchanger parameters and the effects of fins. In general, fins introduce a penalty pressure drop, but, on the other hand, the increase in the Nusselt number is more expressive, and this is the reason for its use. To further improve heat exchange equipment, some recent work demonstrates that, in addition to the fins, devices called vortex generators (VGs) can be employed to increase heat exchange, as discussed by Zdanski, Pauli and Dauner (2015), Salviano, Dezan and Yanagihara (2015), Salviano, Dezan and Yanagihara (2016), Samadifar and Toghraie (2018) and Zanatta et al. (2020), who developed studies that shows increase in heat transfer and pressure drop penalty.

Within the framework of technique development, some authors addressed ways to calculate the convective heat transfer coefficient, such as Chen, Chou and Wang (2007), Huang, Yuan and Ay (2009) and González, Vaz Jr. and Zdanski (2019), who developed methodologies using hybrid approximations, employing experimental data and numerical methods to calculate the convective heat transfer. Particularly, the technique proposed by González, Vaz Jr. and Zdanski (2019) is interesting, because the fin geometry is not an obstacle, and, in this methodology, a hybrid numerical-experimental approach was presented to obtain the thermal performance of heat exchangers.

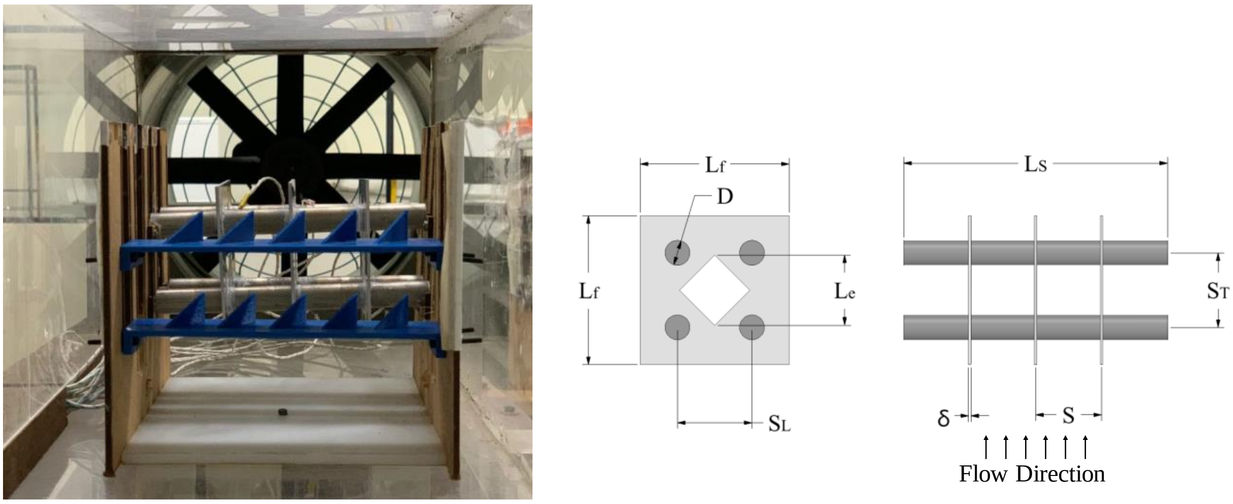
The present study unites the concepts already studied about vortex generators by Zdanski, Pauli and Dauner (2015) and Zanatta et al. (2020), as well as the experimental numerical method proposed by González, Vaz Jr. and Zdanski (2019), so that the effects of vortex generators and thermal grease in finned tube heat exchangers devices are measured, for different fin geometries.

2. NUMERICAL-EXPERIMENTAL METHODOLOGY

The methodology used in this paper is a numerical-experimental hybrid method, with experimental data used in the simulations as boundary conditions. The commercial software Ansys-Fluent® was used for the numerical simulations, and a wind tunnel for the experimental procedure.

2.1 Experimental procedure

The experimental tests were divided in four rounds, with and without vortex generators (VG), and with and without adding thermal grease, changing the flow velocity for each round, resulting in 16 experiments. The apparatus utilized in the experiments were previously built and tested by Rauber (2021) and Rauber et al. (2022), and consists of three aluminum plate fins evenly spaced in four stainless steel heaters. The vortex generator, when added, is placed in front of the heat exchanger and parallel to the heaters, according to Fig. 1(a), in this figure the setup of thermocouples can be seen. The geometry and dimensional parameters of the heat exchanger are presented in Fig. 1(b) and the parameter values in Tab. 1. The parameters used for the vortex generation configuration is shown in Fig. 2 and its dimensional values in Tab. 2.



(a) (b)

Figure 1. Heat exchanger (a) Assembly of the apparatus and (b) dimensional measures.

Table 1. Dimensional parameters for heat exchanger from Rauber et al. (2022) work.

Parameter	Symbol	Value
Fin spacing	S	42.50 mm
Fin length	L_f	96.00 mm
Tube diameter	D	16.0 mm
Tube length	L_s	170.0 mm
Fin thickness	δ	1.50 mm
Longitudinal pitch	S_L	48.0 mm
Transversal pitch	S_T	48.0 mm
Perforation length	L_e	48.0 mm
Number of fins	N_f	3
Number of tubes	N_s	4

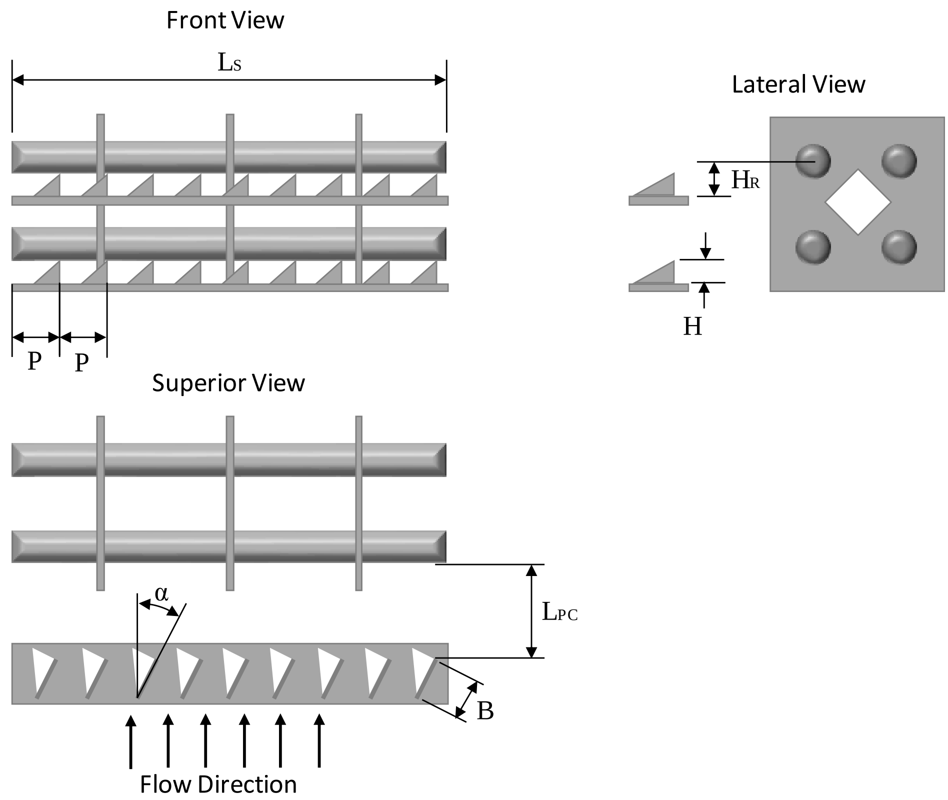


Figure 2. Parameters of the VGs and its assembly in the holding support (Zanatta et al., 2020).

Table 2. Geometric parameters of the vortex generators.

Parameter	Values
P/D	1.75
L_{PC}/D	1.94
$\alpha\ [rad]$	0.72
B/D	1.75
H/D	1.00
S_T/D	3.00
S_L/D	3.00
H_R/D	1.00

The contact between the fins and heaters is through pressure (interference assembly), in which 8 tests were performed with pressure contact only and 8 tests with a layer of thermal grease in the base of the fin contact with the heater, as shown in Fig. 3. Table 3 presents the nomenclature used for define the assembly of the apparatus.

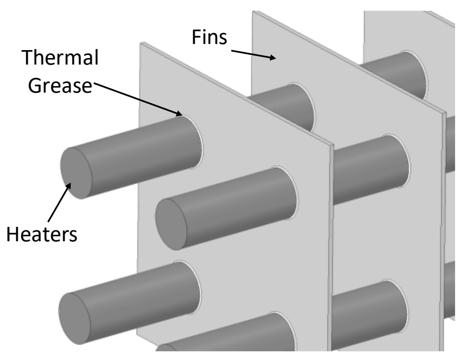


Figure 3. Layer of thermal grease in heat exchanger apparatus.

Table 3. Nomenclature used for each assembly of the apparatus.

	Thermal Grease	Vortex Generator	Named
Without TG and VG	no	no	A
Without TG with VG	no	yes	B
With TG without VG	yes	no	C
With TG with VG	yes	yes	D

The wind tunnel has a squared test section, named A_{ts}, of $250.0 \times 250.0\ mm^2$ and operates by suction with ratio of $1:6$. The works of Bender et al. (2018), González, Vaz Júnior and Zdanski (2019), Zanatta et al. (2020), Rauber (2021) and Rauber et al. (2022) were performed in the same wind tunnel in studies involving heat exchange and aerodynamics. According to Zdanski, Pauli and Dauner (2015), the intensity of the flow turbulence entering the empty test section is less than 1%. The scheme of the wind tunnel tests is depicted in Fig. 4.

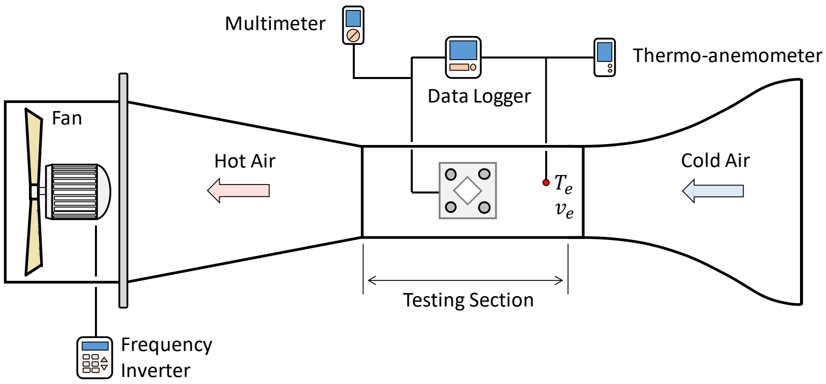


Figure 4. Experimental scheme.

The superficial temperatures of the heaters are obtained using a Data Logger that read the measurements of 12 type K thermocouples, three welded to each heater in order to obtain a well distributed result. The room temperature and flow velocity are measured with a thermo-anemometer, the electrical resistance of the heaters and the input voltage through a digital multimeter.

2.2 Numerical procedure

The numerical simulation was performed with the help of Ansys-Fluent® commercial package. The numerical-experimental method proposed by González, Vaz Júnior and Zdanski (2019) was adopted, in which, after an iterative calculation, it is possible to obtain the average heat transfer coefficient, $\bar{h}_0$, and the overall efficiency of the finned surface, $\bar{\eta}_0$. The numerical part of the method involves simulation of the heat conduction problem in the fin using the average temperature of the heaters as a boundary conditions, with the average heat transfer coefficient and the room temperature. The surfaces in the thickness of the fin are considered isothermal.

However, to perform the iterative method, it is necessary to follow the steps:

1 – Dimensional measurements of the cross-sectional and total heat transfer area of the finned surface, where 'N_f' is the number of fins, 'N_s' is the number of heaters, 'A_s' is the surface area of one heater and 'A_f' is the surface area of one fin, with this the total area of the heat exchange is,

$$A_{total} = N_f A_f + N_s A_s \ ; \tag{1}$$

2 – Measurement of the air flow velocity, v_∞, room temperature, T_∞, average temperature of the heaters, T_s calculated according to Eq. 2, where 'T_i' is a thermocouple temperature. The electrical voltage, 'V', and electrical resistance of the heaters, 'R_j';

$$T_s = \frac{1}{3N_s} \sum_{i=1}^{3N_s} T_i \ ; \tag{2}$$

3 – Evaluation of the film temperature and thermodynamic properties of the air to calculate the Nusselt number and Reynolds number,

$$T_{film} = \frac{T_s + T_\infty}{2} \; ; \tag{3}$$

4 – Computation of the experimental total heat transfer rate, '$\dot{Q}_{exp}$',

$$\dot{Q}_{exp} = \cos(\phi)\, V^2 \sum_{j=1}^{N_s} \frac{1}{R_j} \; ; \tag{4}$$

5 – Calculation of the tunnel exit temperature, 'T_{out}', using the principle of energy conservation in conjunction with the heat transfer rate (this procedure is used by González, Vaz Jr. and Zdanski (2019)),

$$T_{out} = \frac{\dot{Q}_{exp}}{\rho v_\infty A_{ts} C_p} + T_\infty \tag{5}$$

6 – Evaluation of the global heat transfer coefficient, 'U', using the Newton's law of cooling,

$$U = \frac{\dot{Q}_{exp}}{A_{total} \Delta T_{lm}} \quad \text{where } \Delta T_{lm} = \frac{(T_s - T_\infty) - (T_s - T_{out})}{\ln\left(\frac{(T_s - T_\infty)}{(T_s - T_{out})}\right)} \tag{6}$$

7 – Computation of the average convective heat transfer coefficient and the overall efficiency of finned surface using the following iterative procedure:

i – Arbitrary choice of average heat transfer coefficient value '$\bar{h}_0$';
WHILE $\varphi_h > TOL_h$ *DO:*
Calculate the heat transferred over the surface of all cylindrical heaters,

$$\dot{Q}_s = \bar{h}_0 N_s A_s \Delta T_{lm} \; ; \tag{7}$$

ii – Determine numerically the heat flux transferred by the surface of one fin, 'q_f', using the average experimental temperature of the heater, T_s, with a boundary condition in the simulation, and the average convective heat coefficient, $\bar{h}_0$;

iii – Calculate the total heat exchanged by the fins and heaters

$$\dot{Q}_{total} = N_f A_f q_f + \dot{Q}_s \; ; \tag{8}$$

iv – Calculate the maximum ideal heat exchanged

$$\dot{Q}_{max} = \bar{h}_0 A_{total} \Delta T_{lm} \; ; \tag{9}$$

v – Get the overall efficiency of the finned surface,

$$\bar{\eta}_0 = \frac{\dot{Q}_{total}}{\dot{Q}_{max}} \; ; \tag{10}$$

vi – Obtain a new value for the average heat transfer coefficient,

$$\bar{h}_0 = \frac{U}{\bar{\eta}_0} \; ; \tag{11}$$

vii – Evaluate the convergence index of the average heat transfer coefficients obtained,

$$\varphi_h = \left|\left(\bar{h}_0\right)_{new} - \left(\bar{h}_0\right)_{old}\right| \quad ; \tag{12}$$

END WHILE

viii – Output values ($\bar{h}_0$, $\bar{\eta}_0$ and $\dot{Q}_f$) converged.

To calculate the Nusselt number and the Reynolds number, the correlation presented by Frass (2015) for the hydraulic diameter was used. The hydraulic diameter is calculated according:

$$Dh = \frac{4V'\Psi}{A'} \quad \text{where} \quad \begin{aligned} V' &= S_T S_L (S-\delta) \\ A' &= 2(S_T S_L - \pi D^2/4) + \pi D (S-\delta) \\ \Psi &= 1 - \delta/S - (\pi D^2/4)\,(S-\delta)/(S_T S_L S) \end{aligned} \quad . \tag{13}$$

The Reynolds number for the comparisons is calculated based on:

$$Re_{Dh} = \frac{\rho_a \, u_{max} \, Dh}{\mu_a} \ , \tag{14}$$

where ρ_a and μ_a are the specific mass and dynamic viscosity of air, respectively, and u_{max} is defined by: $u_{max} = {}^{u_\infty}/_{\Psi}$. For the comparisons between the experimental data and the literature correlation were made using Kaminski and Gross equation, explained for Frass (2015). The Nusselt's correlation is:

$$Nu_{Dh} = C_1 \, Re_{Dh}^{0.625} \, {Pr_a}^{1/3} \left(\frac{Dh}{S}\right)^{1/3} \ , \tag{15}$$

in which Pr_a is the air Prandtl number and $C_1 = 0.25$ for two rows and inline arrangement. In the above equations all measures for the calculations are in the Fig. 1(b) and Tab. 1.

3. RESULTS AND DISCUSSIONS

To validate the simulation results, firstly the size of the elements is varied to find the convergence and define the size of the mesh for the rest of the simulations. Then, the Nusselt number is compared with results from the literature.

In the Fig. 5(a), the mesh refining convergence is presented with respect to the Nusselt number (based on average convective coefficient, $\bar{h}_0$). It was observed that the number of elements in the mesh, using elements with 1.0 mm are already less than 0.1% different from elements with 2.0 mm, so the mesh size is chosen to be with 1.0 mm elements only by display quality of the temperature profile. Figure 5(b) shows the convergence of the Nusselt number using the González, Vaz Jr. and Zdanski (2019) method presented in section 2.2 for the mesh with element size of 1.0 mm. Both validation cases of the simulation were performed for the first Reynolds number of the fin case C from Tab. 3, which is the same case validated in Rauber's (2021) work.

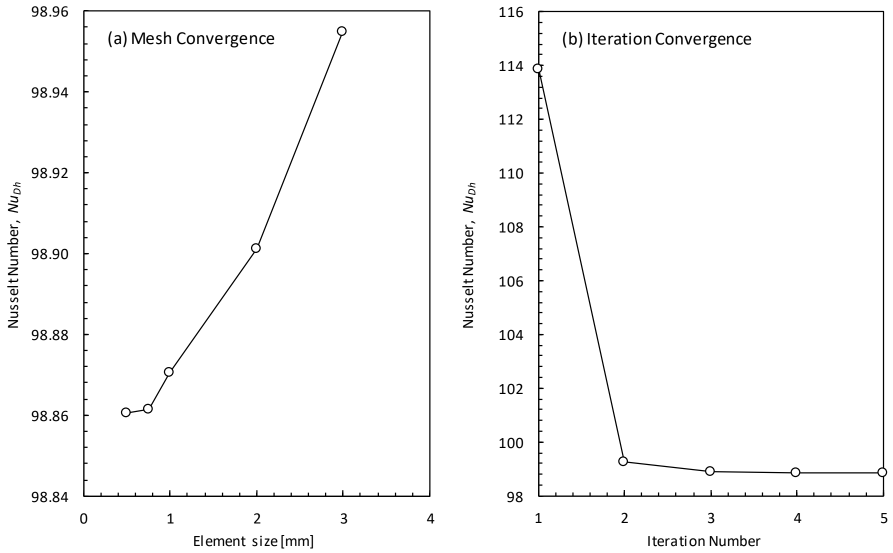


Figure 5. Convergence tests. (a) Mesh convergence and (b) Iteration convergence

Figure 6(a) shows the fin model used in the simulations with the mesh designed with Ansys-Mesh® and Fig. 6(b) shows the temperature profile simulated. This figure shows the region where the boundary condition of T_s is applied internally in the heaters holes.

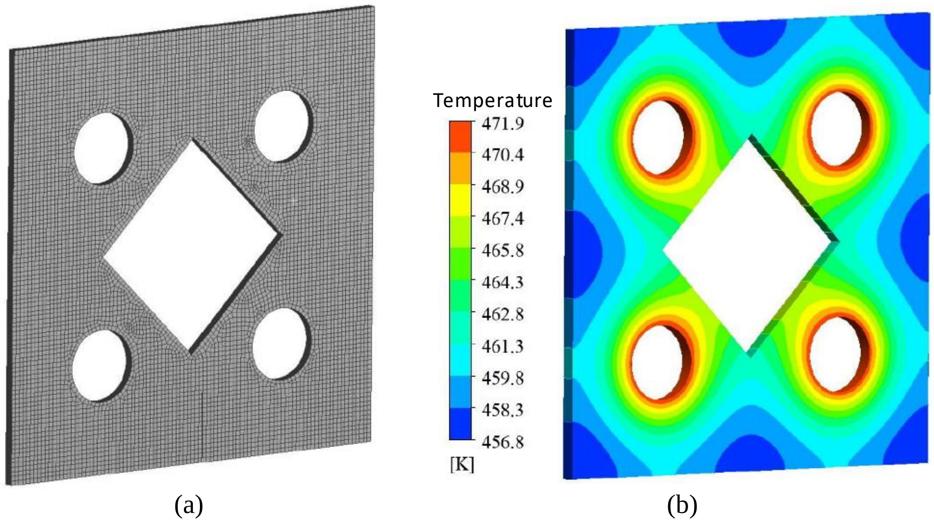


Figure 6. Simulated fin (a) Mesh with 1,0 mm sized elements (b) Temperature distribution.

With the mesh defined, all the others simulations are done and the Nusselt number is used as a comparison with the experiments done by Rauber (2021) for the same fin (Case C from Tab. 3), as shown on Fig. 7. The mean deviation between the two curves is 2.37% and the maximum is 3.12% for the Nusselt number. With these results, it is concluded that the methodology is valid and will produce reliable results.

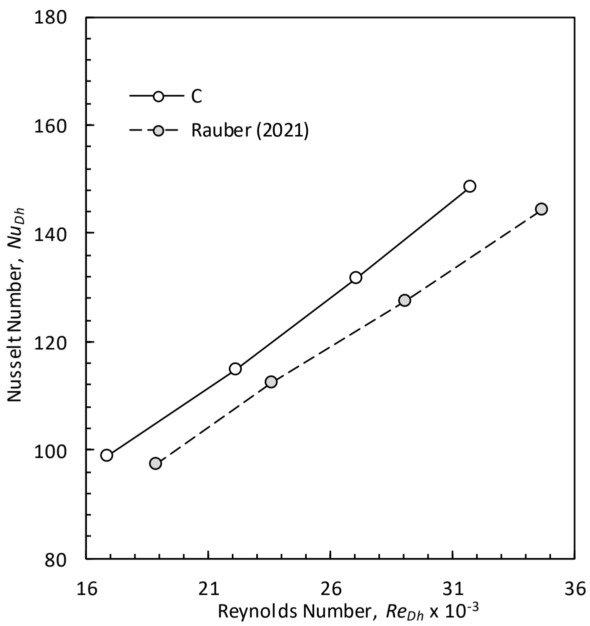


Figure 7. Nusselt number comparative with the literature.

3.1 Comparisons between configurations

In order to compare the different experimental setups, graphics were prepared to analyze the Nusselt number as a function of the Reynolds number. It is expected that the systems with the vortex generator will result in higher Nusselt number, meaning that, when the flow is more turbulent, there is a higher heat transfer rate. Also, it is expected that the systems without the thermal grease will exchange less heat, meaning that the thermal grease acts as an agent that dissipates heat, helping the layer of contact between heaters and fins to exchange more heat. Without thermal paste, the space between heaters and fins can form a pocket of air, which acts as an insulator and prevents heat from the heaters to be exchanged to the fins and dissipate from there. The results for these comparisons are shown in Fig. 8.

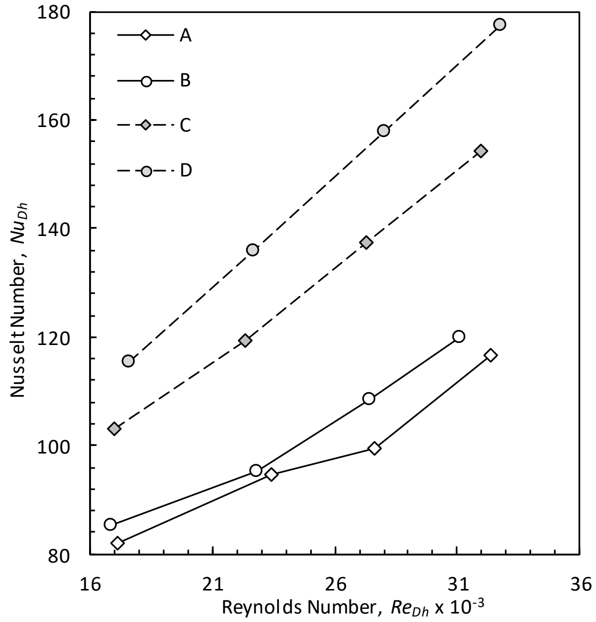


Figure 8. Comparison between systems.

As expected, Fig. 8 proves the hypothesis assumed in the beginning. The concepts of turbulent flow and the consequences toward heat exchanges as presented by Zanatta et al. (2020) are proven when the systems with the addition of the vortex generators exchange more heat and thus show higher Nusselt numbers. The addition of thermal grease was proved to enhance the thermal flow between heaters and fins.

Table 4 shows the improvement on the Nusselt number between the systems, using the nomenclature from the Tab. 3. It is clear that, when the thermal grease is added, use of vortex generators improve performance, because the heat exchange between heaters and fins is more efficient, so the difference between the systems with and without the vortex generator is more significant. When there is no thermal grease, the pressure contact only cannot guarantee the best heat exchange between heaters and fins, so the vortex generators, although still making a difference, are not as significant for these systems. Still, when the vortex generator is added, both systems respond as expected and show an improvement in the Nusselt number.

Table 4. Average Nusselt number improvement between systems (nomenclature from Tab. 3).

Comparison between systems	Relative absolute difference
C-D	20.2%
A-B	3.9%
B-D	29.9%
A-C	23.3%

4. CONCLUSIONS

The objective of this study was to compare four different systems for their Nusselt number. The systems consisted of the same four heaters and three fins, two were with an added vortex generator and two were without the addition of thermal grease to the connection between heaters and fins. All systems were tested in a wind tunnel, in order to use the experimental data as boundary conditions to simulate the heat flux in the fins with the support of the commercial software Ansys-Fluent®. The simulations provide the convective coefficient (based on an iterative procedure) used to calculate the Nusselt number.

For the experimental procedure, four different velocities were used in the wind tunnel, and five simulations were necessary to converge the results for each system, thus resulting in 80 simulations in total, aside from the 25 simulations run in the beginning to define the necessary refinement for the mesh.

Firstly, a convergence analysis was performed to determine the necessary refinement for the mesh used in the simulations, as well as the number of simulations necessary to converge to reliable results for each system. After those, the first system is simulated and compared to the results presented in Rauber (2021), who adopted the same heat exchanger used in this paper, with the same perforated fins and the same hybrid methodology for the analysis. Once the results were validated, the other four systems were simulated in order to compare the results between each other.

With the four systems experimented and simulated, the Nusselt number *versus* Reynolds number was plotted, in order to keep the same basis of comparisons for all the systems. The difference between the average of the system with thermal grease with and without the vortex generator is 20%, for the system without thermal grease with and without the vortex generator is 4%. Between the systems without the vortex generator with and without thermal grease is 23% and with the vortex generator with and without thermal paste is 30%.

The results presented in this paper prove the hypothesis initially assumed by the authors, i.e. addition of vortex generators results in the improvement of the heat exchange rate (higher Nusselt number), and use the thermal grease causes also an increase of the heat transfer rate. Noticeably, the systems without the addition of thermal paste proved the existence of air pockets between the fins and the heaters, which acts as insulators and diminish the heat exchange between heaters and fins, leading to smaller Nusselt numbers.

5. ACKNOWLEDGEMENTS

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6. REFERENCES

Bender, A., Meier, A.M., Vaz Jr, M., Zdanski, P.S.B., 2018. “A numerical study of forced convection in a new trapezoidal tube bank arrangement*”. International Communications in Heat and Mass Transfer*, Vol. 91, p. 117-124.

Chen, H.-T., Chou, J.-C., Wang, H.-C., 2007. “Estimation of heat transfer coefficient on the vertical plate fin of finned-tube heat exchangers for various air speed and fin spacings”. *International Journal of Heat and Mass Transfer*, Vol. 50, p. 45-57.

Choi, J.M., Kim, Y., Lee, M., Kim, Y., 2010. “Air side heat transfer coefficients of discrete plate finned-tube heat exchangers with large fin pitch”. *Applied Thermal Engineering*, Vol. 30, p. 174-180.

Frass, F., 2015. *Principle of Finned-Tube Heat Exchanger Design for Enhanced Heat Transfer*. WSEAS Press, Vienna, 2nd edition.

González, A.M., Vaz Jr., M., Zdanski, P.S.B., 2019. “A hybrid numerical-experimental analysis of heat transfer by forced convection in plate-finned heat exchangers”. *Applied Thermal Engineering*, Vol. 148, p. 363-370.

Huang, C.H., Yuan, I.C., Ay, H., 2009. “An experimental study in determining the local heat transfer coefficients for the plate finned-tube heat exchangers”. *International Journal of Heat and Mass Transfer*, Vol. 52, p. 4883–4893.

Rauber, W.K., 2021. *Numerical-Experimental Evaluation of the Performance of Finned Heat Exchanger: Study of Geometric Configuration for the Fins*, original in Portuguese “*Avaliação numérico-experimental do desempenho de trocadores de calor aletados: estudo de configurações geométricas para as aletas*”. Master’s thesis, Santa Catarina State University, Brazil.

Rauber, W.K., Silva, U.F., Vaz Jr, M., Alves, M.V.C, Zdanski, P.S.B., 2022. “Investigation of the effects of fin perforations on the thermal-hydraulic performance of Plate-Finned heat exchangers”. *International Journal of Heat and Mass Transfer*, Vol. 187, p. 122561.

Salviano, L. O., Dezan, D. J., Yanagihara, J. I., 2015. “Optimization of winglet-type vortex generator position and angles in plate-fin compact heat exchanger: Response Surface Methodology and Direct Optimization”. *International Journal of Heat and Mass Transfer*, Vol. 82, p. 373-387.

Salviano, L. O., Dezan, D. J., Yanagihara, J. I., 2016. “Thermal-Hydraulic performance optimization of inline and staggered fin-tube compact heat exchangers applying longitudinal vortex generators”. *Applied Thermal Engineering*, Vol. 95, p. 311-329.

Samadifar, M., Toghraie, D., 2018. “Numerical simulation of heat transfer enhancement in a plate-fin heat exchanger using a new type of vortex generators”. *Applied Thermal Engineering*, Vol. 133, p. 671-681.

Wang, Q., Qian, Z., Cheng, J., Huang, W., Ren, J., 2019. “Analysis on thermal hydraulic performance of the elliptical tube in the finned-tube heat exchanger by new method”. *International Journal of Heat and Mass Transfer*, Vol. 134, p. 388-397.

Zanatta, V.N., Silva, U.F., Rauber, W.K., Vaz Jr, M., Zdanski, P.S.B., 2020. “A hybrid numerical-experimental analysis of the effects of delta winglet vortex generators in plate-finned heat exchangers with forced convection”. In *Proceedings of the 12th Spring School Transition and Turbulence, EPTT2020*. Blumenau, Brazil.

Zdanski, P.S.B., Pauli, D., Dauner, F.A.L., 2015. "Effects of delta winglet vortex generators on flow of air over in-line tube bank: A new empirical correlation for heat transfer prediction". *International Communications in Heat and Mass Transfer*, Vol. 67, p. 89-96.

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